TECHNICAL MEMORANDUMS

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

No. 403

HIGH-SPEED OIL ENGINES FOR VEHICLES

By Ludwig Hausfelder

PART II

From "Der Motorwagen"
September 30, and December 10, 1926

Washington
March, 1927

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS.

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PART II.*

Further progress toward the satisfactory solution of the difficult problem of the distribution and atomization of the injected fuel was made by extensive experimentation with various fuel valves, nozzles and atomizing devices. Valuable information was also obtained through numerous experimental researches on the combustion of oils and the manner of introducing the combustion air into the cylinder, as well as on the physical processes of atomization, the determination of the size of the drops, etc. These researches led to the conclusion that it is possible, even without producing great turbulence in the combustion chamber and at a moderate pump pressure, if the degree of atomization and the penetrative power of the fuel jet are adapted to the shape of the combustion chamber and to the dimensions of the cylinder ("Zeitschrift des Vereins deutscher Ingenieure," 1925, p.1261). In order to prevent the fuel jets from striking against the piston and cylinder walls, the Deutz and the Augsburg engine companies both employed concave piston heads and so regulated the penetrative

*From "Der Motorwagen," of September 30, (pp. 649-654) and December 10, (pp. 841-850), 1926.

power of the fuel jets by means of suitably shaped nozzles, that the oil drops were burned before reaching the piston head (Fig. 11). In this way very good combustion was obtained at moderate compression pressures, although accompanied, at the piston dead center, by a pressure increase approaching explosion and by correspondingly high working stresses.

The Krupp Diesel engine (Fig. 12) works at approximately constant combustion pressure. In contrast with the above-mentioned engines, the combustion space is relatively flat in this engine. Not only is there no attempt to prevent the jet from striking the walls, but it is made to do so by elevating the middle of the piston head so that the angle of incidence of the fuel jets, which at first are nearly tangential to the surface, gradually increases during the injection. The Krupp Company based this method on the assumption that the conical fuel jet is dispersed by striking the piston head and is thus distributed throughout the combustion chamber in the form of a fine mist. It is hard to say whether this assumption is correct. Fuel drops with a diameter of 20-30 μ already have such a surface tension that any further atomization would hardly be effected by the nearly tangential contact. It is more reasonable to assume that the central portion of the piston head remains very hot and therefore vaporizes, through the effect of heat alone, the fuel drops which strike it at a lower velocity. The efficiency of this type of engine is largely due to

giving the combustion air a circular motion, as already proposed by Hesselman (N.A.C.A. Technical Memorandum No. 312). This circular motion of the air around the cylinder axis is produced by a directing surface at the intake valve. The circular motion produced by the intake stroke continues to the end of the compression stroke and even during combustion, as demonstrated by experiments ("V.D.I.," 1925, p.673). It is worth noting in this connection that the efficiency of the combustion bears a quite definite relation to the velocity of the circulating air, as demonstrated by Hintz in his lecture at the 1925 annual meeting of the Association of German Engineers in Augsburg.

As regards stationary engines, the mechanical and ignition-chamber injection methods have proved equally successful. If the former method is more efficient with respect to economy of fuel and varying load, the ignition-chamber engine is less sensitive and easier to tend, on account of its lower pump pressure and lower maximum combustion pressure.

We must now consider as to which of the two systems is better adapted for high-speed vehicle engines. In this connection I wish to eliminate the annular-whirl method as unsuitable (at least as it has hitherto been used in so-called "displacer engines"), due to the low pressure and coarse atomization, and because the desired change in the ignition timing is not possible under the variable operating conditions of vehicle engines. On the other hand, a combination of the annular-whirl method

and the mechanical-injection method can be advantageously used, as will be more fully explained farther on.

As for the ignition-chamber method, this has, first of all, the exceptional advantage of enabling uniform functioning and smooth transition from compression to ignition and expansion.

The low maximum pressures render it possible to make the driving parts relatively light. Moreover, the ignition-chamber method enables lower pump pressures and correspondingly larger nozzle openings, a circumstance which is very valuable for the reliability of an engine, on account of the impurities in the fuel, which are unavoidable even with the best filtering. On the other hand, it is more difficult to regulate, due in part to the fact that the walls of the ignition chamber tend to become coated with oil carbon, which absorbs a portion of the fuel and causes after-combustion. The endeavor to diminish the heat-conducting walls of the combustion chamber as much as practicable leads to making the ignition chamber as small as possible. This is accompanied by the disadvantage, however, that it can be but poorly cleaned out and still contains combustion residues after the charging of the cylinder with air whose small residue of oxygen is often insufficient to insure ignition. This applies, however, only to engines in which the fuel, coarsely atomized in the ignition chamber, is injected through narrow channels into the combustion chamber by the

partial combustion with considerable excess pressure. Recently a modification has been introduced in the Diesel engines of De La Vergne Engine Company and of the Falk Corporation by making relatively larger passages between the ignition chamber and combustion chamber, so that there is no great pressure difference between the two. These engines, which in general employ the Price method of fuel injection (Fig. 13), are characterized by the fact that the combustion space, which is connected with the working cylinder by a wide round or rectangular neck, is divided into two chambers, each with a fuel nozzle, whose axes are inclined to the cylinder axis ("V.D.I.," 1925, p.1084). The shape of these chambers (conical or rectangular) is adapted to the shape of the fuel jet (conical or fan-shaped). The claim that a supplementary atomization is produced by the impact of the two fuel jets can hardly be maintained, in view of the great stability of very small drops. The success of these engines is due rather to the good adaptation of the fuel jet to the combustion chamber, as well as to the air turbulence caused by this manner of construction. Of course the enlarging of the connecting passage or neck necessitates a finer atomization in the ignition chamber. Instead of projecting unburned oil drops into the combustion chamber to be burned there, by far the greater portion of the injected fuel is already burned in the ignition chamber. The air entering the combustion chamber, under violent agitation produced by the constriction, pro-

duces a rapid and complete vaporization of the fuel drops, as well as a rapid propagation of the ignition waves and the introduction of new portions of air into the combustion zone. The good experimental results, which were recently obtained (according to an American communication, the Banner engine of the Falk Corporation attained a mean pressure of 156 lb./sq.in.), have sufficiently demonstrated that the motion of the air which propagates the combustion in the combustion chamber is, under certain conditions, more important than to have very compact combustion chambers with small wall areas in contact with the flames. As an analogous case, reference is here made to the experiments of Riccardo (The Internal Combustion Engine, Vol. II, p.88 ff.), who demonstrated that, even with explosion engines having laterally located valves, i.e., with relatively large heat-conducting wall areas, it is entirely possible to obtain excellent combustion conditions by the favorable utilization of the whirling motion of the gaseous mixture.

Although it is possible to build engines suitable for vehicles on the ignition-chamber principle, I nevertheless believe that in future the mechanical-injection method will be preferred. As contrasted with the ignition-chamber method, there is here only a single atomization and combustion process, whereby the relations are clearer and easier to control. Hereby, the same as in the stationary hollow-piston engines of Deutz and the M.A.N., the fuel jet is given such a direction

and penetrative force, that it burns without impinging on the piston or cylinder walls. Any contact with the metal walls, especially when they are coated with lubricating oil, is to be avoided, not only on account of the direct heat transmission, but also on account of the catalytic action of the wall material, which accelerates the pyrogenic decomposition of the fuel. The same is true, in a still greater degree, of the contact with the oil carbon deposited on the piston head, since this absorbs the fuel, causes misfires and after-burning, and greatly damps the control. The modern vehicle engine seems to be developing within the limit represented by these fiducial lines. The fact that the problem has not yet been fully solved, is due to the extraordinary difficulties arising when one tries to reach the goal simply by making slight changes in the dimensions of stationary engines. The successful development of the high-speed engine must be based rather on a thorough investigation of the whole process, whose individual phases must fully conform to the changed conditions.

For the obtention of high revolution speeds, i.e., high piston and combustion speeds, two conditions are requisite, namely, quick ignition of the injected fuel and rapid combustion. As regards the ignition, we know that the fuel drops injected into the cylinder receive heat by conduction from the hot compressed air. The process is very similar to the combustion process in the coal-dust furnace investigated by Nusselt.

According to Riehm ("V.D.I.," 1924, p.644), the simultaneously occurring heat absorption by radiation can be disregarded, since the gas and wall temperatures are relatively low from the beginning of the combustion. The heat absorbed by conduction is

$$4\,r\,\pi\,\lambda\,(t_L - t) \tag{1}$$

in which

r = radius of drop (m),

λ = coefficient of heat conductivity of air ($kcal\ m^{-1}\ h^{-1}\ {}^0C^{-1}$),

t_L = compression temperature of the working air (^oC),

t = temperature of fuel drop (^oC),

t_a = initial temperature of fuel drop (^oC),

c = specific heat of liquid ($kcal\ kg^{-1}\ {}^oC^{-1}$),

γ = density of liquid ($kg\ m^{-3}$),

z = ignition time of drop after jet leaves nozzle (h).

The absorbed heat raises the temperature of the fuel drops so that

$$4\,r\,\pi\,\lambda\,(t_L - t) = \frac{4}{3}\,r^3\,\pi\,c\,\gamma\,\frac{dt}{dz} \tag{1a}$$

or

$$dz = \frac{r^2\,c\,\gamma}{3\,\lambda}\,\frac{dt}{t_L - t} \tag{1b}$$

For the limiting condition $t = t_a$ for $z = 0$ the integration of this equation gives

$$z = \frac{r^2 c \gamma}{3 \lambda} - \ln \frac{t_L - t}{t_L - t_a} \qquad (2)$$

From this equation the time z, required to reach a temperature t of the fuel drop, can be computed.

The question as to what temperature is necessary for starting the ignition, cannot be answered with certainty, so long as we do not know whether a substantial portion of the fuel is changed into the form of vapor or gas shortly before the ignition of the drop, or whether the drop burns in the liquid form. Opinions differ on this point and no complete answer can be expected very soon, owing to the difficulty of investigating the phenomena experimentally. Probably no considerable amount of gas or vapor is formed before ignition occurs, because the available time (about 0.004 sec. in high-speed engines) is too short and the temperature at the instant of ignition is too low. On the other hand, an envelope of vapor and air will be formed on the surface of the drop by the partial vaporization of the more volatile components of the fuel. This envelope has the lowest possible ignition temperature and consequently ignites first, producing a sudden increase in temperature of the whole drop which then burns in the liquid form. Since, in this case, almost all the heat imparted to the drop is liquid heat, the ignition point of the liquid itself can be unhesitatingly put for t in equation (2). Even if this assumption

were incorrect and a considerable portion of the drop evaporated previously, the correctness of the numerical values calculated from the equation would be but slightly affected. Regardless of whether the heat imparted to the fuel drop is chiefly liquid heat or partially vaporization heat, the above formula shows the great effect of the compression temperature on the ignition instant. The attainment of a high air temperature is therefore an important means for shortening the ignition delay. The size of the drop is still more important, however, its effect on the ignition speed being to increase z according to the square of the drop radius. The combined effect of both factors is shown in Table I, which gives the ignition delays for drops of various sizes at different air and ignition temperatures, on the assumption that there is no previous vapor formation.

Table I.

Ignition delay in seconds $\times 10^{-1}$.

$c = 0.45$ kcal kg^{-1} $^{\circ}$C^{-1},

$\gamma = 900$ kg m^{-3},

$\lambda = 0.0405;\ 0.0425;\ 0.0448$ kcal m^{-1} h^{-1} $^{\circ}$C^{-1}

at $t = 500^{\circ};\ 550^{\circ};\ 600^{\circ}$C.

Ignition temperature	Air temperature	Size of drop in μ = 0.001 mm		
		50 u	25 u	10 u
200°	500	141	35.3	5.65
	550	119	29.7	4.75
	600	102	25.5	4.08
300°	500	262.5	65.6	10.5
	550	215	53.5	8.6
	600	179	45.0	7.15
400°	500	472.5	118	18.9
	550	362.5	90.5	14.5
	600	290	72.5	11.6
500°	500	–	–	–
	550	675	167	27
	600	481	120	19.3

Rapid combustion of the injected fuel is just as important as quick ignition. According to the Wenzel law, the rapidity of the combustion, like the speed of any other chemical reaction in dispersion systems, is proportional to the area of the reacting surface. The best way to accelerate the combustion is therefore the obtention of the greatest possible reaction surface of the total injected fuel, i.e., the formation of the largest possible number of fuel drops of the smallest possible diameter. The direction to follow is plainly indicated since, with decreasing size, the volume of every fuel drop decreases

with the third power, but the surface area only with the second power.

Aside from the chemical process, the rapidity of the combustion also depends on the physical processes and the diffusion of the oxygen. Around every drop an envelope is first formed out of the combustion products H_2O and CO_2, through which the fresh air must pass. A rapid diffusion is favored by a minimum size of the drops, as well as by a violent relative motion of the fuel and air. This last requirement brings us directly to another important condition, namely, the most uniform possible diffusion of the fuel in the combustion air.

Though the formation of a homogeneous mixture of air and the vapors of the easily ignitible fuels offers no great difficulty in explosion engines, a like condition is not easily effected in Diesel engines. What effect, however, the finest possible diffusion of the fuel in the combustion air has on the whole combustion process has been shown by Haber's experiments ("Zeitschrift für angewandte Chemie," 1923, p.661), which led to the important conclusion that, in the combustion of oil vapors with theoretically sufficient air, much CO is always formed instead of CO_2. The reason for this is to be sought in the fact that liquid sprays, especially when the size of the suspended drops varies greatly, are much less uniformly distributed than a mixture of gases and vapors. Their ignition, as likewise the propagation of the combustion, is therefore

slower and requires a greater excess of air. All efforts to obtain rapid combustion with only a small excess of air or, in other words, high revolution speeds with small cylinder dimensions, must therefore culminate in the effort to produce the most homogeneous possible mixture of very finely atomized fuel and air and to burn it very quickly.

There have been only a few theoretical investigations of the nature of the atomization of a fuel jet injected at a high velocity, and we are dependent on the work of hydrodynamic engineers who have experimentally investigated the outflow and turbulence problem with water ("Zeitschrift für angewandte Mathematik und Mechanik," 1921, p.436). Kuehn investigated the nature of atomization in his article "Atomization of Liquid Fuels" (N.A.C.A. Technical Memorandum No. 331), but his attention was devoted more to the determination of the size of the fuel drops than to the real atomization problem. Very recently Triebnigg has attempted to solve the problem of atomization and determine the physical foundations of the injection process on the basis of the capillary theory ("Der Einblase- und Einspritzvorgang bei Dieselmaschinen," Vienna, 1926).

We know that, up to a certain critical velocity, the flow of liquids through small tubes is laminar, with an approximately parabolic velocity curve over the whole cross section. The upper limit for the laminar flow is given by the Reynolds formula

$$c = \frac{R\,v}{d} \qquad (3)$$

wherein R represents a constant which is dependent on the tubular friction; ν, the kinetic viscosity of the liquid; and d, the diameter of the tube. Above this critical velocity, of about 35 m (115 ft.) per second for fuel oils and customary tubular cross sections, there begins, through viscosity, an increase of velocity in a vortical peripheral layer, which, on leaving the nozzle, causes the elimination of the capillary forces of the jet components. The atomization of the jet is greatly affected by the shape of the nozzle opening, whose peripheral action decidedly affects the splitting up of the jet. As a means for obtaining very fine atomization, it is now sought almost exclusively to increase the peripheral action of the nozzle for given cross-sectional areas by means of the greatest possible extension of the periphery of the nozzle opening and very narrow spraying slots. The question as to whether any further atomization is effected by the friction of the jet (as it leaves the nozzle at a high velocity) against the surrounding compressed air, can be answered negatively (at least for high outflow velocities and small drops) according to recent researches. Lenhardt's experiments on the deformation of water drops at high velocity showed that only drops with diameters of more than 0.5 mm (0.02 in.) were so far deformed by friction with the air as to split in two. Smaller drops could not be deformed on account of their greater surface tension. The possibility of further disintegration of finely

atomized oil drops is all the less probable, since their surface tension is about six times as great as that of the water drops tested by Lenhardt. Even the compression ratio of the air has no effect on the atomization, since the viscosity constant of the air, which is $y = 0.000172$ according to Lenhardt, depends only on the temperature and not on the pressure. The atomization can be affected by the compression only in so far as (through the temperature increase caused by the compression) the size of the drops is increased by the absorption of heat or by the formation of vapor envelopes. It will express itself in a change in its resistance and a corresponding deviation from the computed distance of travel. The reaction of the air on the structure of the whole jet can not, however, be entirely disregarded on this account, since it favors the separation of the closely collected fuel drops as they leave the nozzle and the splitting up of the jet into many smaller jets. If the jet leaves the nozzle in the form of a conical spray (as is, for example, the case with single-hole nozzles), drops are thrown off both internally and externally, especially at the beginning of the spray cone. Atomization pictures therefore almost always show a scattering of very small drops near the axis of the cone, as likewise on the periphery.

The size of the fuel drops is a function of the pressure to which it is almost directly proportional. With increasing pump pressure, the uniformity of the individual fuel drops in-

creases with respect to size. If we start with the assumption that, under otherwise like conditions, the excellence of combustion is a function of the atomization and that, on the other hand, the latter is improved by increasing the pump pressure, it is very natural to test the fuel consumption of an engine in terms of the injection pressure. Such experiments were performed by Heidelberg ("V.D.I.," 1924, p.1047), with the result that the fuel consumption was, in fact, nearly inversely proportional to the increase in the injection pressure, but that, above a certain point, the fuel consumption was not affected by a further increase in this pressure (Fig. 14). Heidelberg offers no explanation for this "constancy range," which differs greatly in extent for the several nozzles tested. Büchner proposed a hypothesis, which, if logically thought out, leads to very interesting results.* He proceeds from the well-known phenomenon, which has been corroborated by experimental investigations on the behavior of air near moving projectiles, that the coefficient of resistance ζ increases rapidly in the vicinity of the velocity of sound, only to fall suddenly after exceeding this velocity. Since he further assumed that the friction of the air also exerts an immediate influence on the atomization, he concludes that this can take place best when the exit velocity of the fuel jet is the same as the velocity

*"Beitrag zu den Grundlagen der schnellaufenden Halbdieselmotoren," Halle, 1926. For translation, see N.A.C.A. Technical Memorandums Nos. 356-358.

of sound. A calculation of the velocity of sound, with the aid of Laplace's formula

$$c = \sqrt{\frac{m\ P}{\gamma}} \qquad (4)$$

(in which P represents the pressure at the nozzle mouth, γ the density of the air and m the exponent of the adiabatic change of state), gives values for c which would correspond to multiples of the injection pressure as measured by Heidelberg. If, on the contrary, we take for granted the outflow velocity corresponding to the best measured fuel consumption and control the exponent m, we then obtain the surprising result that the magnitude of the exponent varies for single-hole nozzles between 0.193 and 0.246, according to the kind of nozzle used. The propagation of the kinetic energy in the hot air can no longer take place adiabatically, when any noteworthy transfer of heat occurs between the hot air and the bearer of the energy, but there will be a polytropic change of state lying far below the isotherm. This view is justified when it is considered that the cold fuel is injected into the hot compressed air and absorbs heat from its environment, the amount of heat being quite large, due to the increase in the coefficient of heat conductivity at high relative velocities. The correctness of Büchner's hypothesis would have to be experimentally verified. At any rate it encourages us to give due consideration to the heat transference between the fuel jet and the compressed air when investigating injection and atomization phenomena.

Since the finest possible atomization is an important, though not essential, condition for rapid and complete combustion, especial attention has always been given to determining the size of the drops. Attempts to determine the size of the drops directly by measuring the dynamic pressure exerted by them on a flat plate placed at different distances from the nozzle were made by Riehm, under conditions claimed to approximate very closely the conditions obtaining in the compression chamber of an engine. In Riehm's experimental apparatus, however, the accuracy of the result was considerably impaired by the fact that the surrounding air was set in motion by the resistance of the fuel drops. There were accordingly developed, in the experimental cylinder, air flows and pressure differences which were also registered on the impact plate. Moreover, the air resistance of the drops varied in a hardly determinable manner, as soon as the relative motion against the air accelerated in the direction of the jet appeared, instead of the absolute motion against the still air.

Extensive researches on the direct determination of the size of the fuel drops were made by Kuehn (N.A.C.A. Technical Memorandum No. 331). His measurements were made by allowing the fuel jet to fall for a very short time on a sooted glass plate by passing through a shutter held directly below the mouth of a spray nozzle. The number of the clearly visible drops on the plate was found by counting and their weight was

found by weighing. This method is very difficult to apply practically (as in testing any given nozzle), since it necessitates complicated apparatus for catching the drops. The weighing is especially difficult, because the various conditions, such as the humidity of the air, temperature, etc., must be determined and the weight correspondingly corrected. Moreover, the size of the drops thus determined is only the mean size and furnishes no basis for the determination of the number and size of the smallest and largest drops and their distribution in the jet picture.

Another method, developed in the laboratory of the "Gesellschaft für Kohlentechnik," at Essen, for determining the size of the drops, has been suggested by Häuser and Strobel ("Zeitschrift für technische Physik," 1924, p.154). Hereby the fuel jet immediately after leaving the nozzle, is caught on a glass plate covered with glycerine. The fuel drops remain longer in suspension in this liquid and can be observed and measured with a microscope. Since the drops sink slowly to the bottom and there run together, the accuracy of the observation is limited. No report has yet been made of comprehensive measurements by this method.

Much more reliable than the above methods is the one of Wöltjen, who, in his dissertation "On the Fineness of the Fuel Atomization in Oil Engines" ("Ueber die Feinheit der Brennstoffzerstäubung in Oelmaschinen," Technical Hochschule, Darmstadt, 1925), proposed an exceptionally practical method for

determining the sizes of the drops. Wöltjen atomized the fuels under conditions as similar as possible to those prevailing in the engine and photographed the drops thus obtained. Starting with the method employed in the chemistry of colloids, the fuel was injected into a receiving liquid in which it remained unchanged for a long time. The dispersing liquid was a mixture of 70% distilled water and 30% "Queol D," a tanning extract of the Colonial Dyestuff Company, Karlsruhe. In order to approximate as closely as possible the conditions obtaining in the engine, the fuel was injected into a steel "bomb," which was partially filled with the dispersing liquid and whose air space was kept, by means of compressed air, under the pressure prevailing in the engine. Since the oil injected into the bomb first passed through the compressed air before entering the dispersing liquid, certain effects of the friction of the air on the size of the drops could be observed. The emulsion of the oil and receiving liquid could then be examined under the microscope with sufficient accuracy, whereby not only the size of the individual drops could be determined, but also the proportionate numbers of the drops of different sizes. Plates I and II show microphotographs of atomization experiments, a-d being photos of atomizations produced by air injection and f-k by airless injection. The pictures plainly show the effect of increasing the injection pressure on the size and uniformity of the drops. For comparison with the artificial emulsions

obtained by oil injection, Plate Ie shows a natural emulsion, namely, cow milk, whose suspended fat globules have about the same size as the finest oil drops obtainable by atomization.

Since the combustion process in an engine and consequently its whole structure are decidedly affected by the choice of a certain size of drops, a critical discussion of the atomization pictures is of interest. The large drops, which are numerous at low pressures, decrease in number with increasing pressure; the number of medium-sized drops increases; the number of the small drops increases still more; and the atomization becomes more uniform. A comparison of Tables II and III shows the different injection pressures required in air-injection and airless-injection engines for producing the same medium-sized drops.

Table II.

Size of drops with air injection.

Plate I. Fig.	Injection pressure p_2 in atm.	Outflow velocity w in m/sec	Drop diameter d_m in 0.001 mm
a	75	372	4.37
b	65	345	13.75
c	50	286	17.50
d	40	220	25.00

Table III.

Size of drops with airless injection.

Plate II. Fig.	Injection pressure p_2 in atm.	Outflow velocity w in m/sec.	Drop diameter d_m in 0.001 mm
g	300	256	4.37
h	250	235	13.75
i	200	210	20.00
j	150	183	26.25
k	100	149	33.75
l	50	112	40.00

It is remarkable that, according to Wöltjen's experiments, the finest atomization (Plate Ia), which produced drops of 4.37 μ diameter at an injection pressure of 75 atm., did not have the anticipated favorable effect in the engine, which showed a tendency to detonate, with the emission of dark-colored exhaust gases. We are therefore led to conclude that, in spite of the considerable energy of the injection air, the finely atomized fuel is not carried far enough into the combustion chamber, but remains immediately in front of the nozzle, in part causing preignitions and in part passing out unburned with the exhaust gases. If the combustion of such small drops meets with difficulties in air injection, still greater difficulties may naturally be anticipated in airless injection. It is certain (as will subsequently be more fully explained) that the distribution of very finely atomized fuel throughout the whole combustion space is possible only with the aid of special auxiliary devices. One method consists in effecting the atomiza-

tion through a suitable nozzle and pressure in such a way as to yield a greater number of coarse drops in proportion to the number of fine drops. The latter will then serve as tinder for starting the combustion, while the former, due to their greater momentum, will penetrate the compression space and propagate the combustion. There will be something more to be said later on the value of this method and its suitability for high-speed vehicle engines.

To return to the faulty combustion in the normal Diesel engine at an air-injection pressure of 75 atmospheres, it should be remarked that it is not necessary to conclude therefrom that drops of the chosen order of magnitude should be unconditionally avoided. The experiment simply shows that the engine was not structurally suited for working at such an injection pressure. I am confident that, through careful adaptation of the size and shape of the combustion chamber, on the one hand, and the suitable guidance of the fuel jets and air, on the other hand, even very finely atomized fuel can be completely burned.

Just as there is a lower limit to the size of the drops, there is also an upper limit. The experiment at an injection pressure of about 40 atm., corresponding to a mean drop diameter of 25 μ (Plate I d) showed that a full loading of the engine was impossible at this low injection pressure. A minimum working pressure of 47 atm. was found to be necessary, at which

the engine worked, however, with a high fuel consumption and very smoky exhaust. This was due to the coarse atomization, because the larger drops did not have time to ignite and burn.

The fact that the experiments described were performed with an air-injection engine, does not affect their importance. It must be assumed that similar experiments with airless-injection engines would give similar results, since they lie within the theoretical limits. The incompleteness of the combustion of very large drops is to be expected on account of the ignition delay undergone by drops of this order of magnitude. Table I shows that, even in normal low-speed Diesel engines with a combustion time of 0.01-0.02 second, drops of 50 μ or more diameter do not have sufficient time for complete combustion, but leave the engine unburned with the exhaust gases.

Formula (2) gives information on the limiting value for the size of the drops which will be ignited within a given time, when it is solved according to the radius of the liquid drop. We then have

$$r_{max} = \sqrt{\frac{3\ \lambda}{c\ \gamma} - \ln \frac{t_L - t_a}{t_L - t}} \qquad (5)$$

For a vehicle engine with a revolution speed of $n = 1200$, let us make the assumption that the ignition delay must not exceed the duration of the injection process. We then have, for an injection period covering a crank angle of 30°, 0.004 sec.

available for ignition and

$$r_{max} = 8.5\ \mu, \quad d_{max} = 17\ \mu.$$

according to equation (5).

That such a size of drop represents only an extreme limit, to be avoided if possible, follows from the consideration that the ignition period of the oil drops last injected would extend over another 30°. Poor combustion with a high final expansion temperature would be the result of this doubtless excessively coarse atomization. It is therefore necessary to restrict the ignition delay at the outset to the smallest possible magnitude, estimated at about 0.0004 sec., corresponding to a crank angle of 3°, and from this to determine the diameter of the drops. In the above example, we then obtain a mean drop diameter of $d = 5.6\ \mu$. Hence this order of magnitude must be maintained for the diameter of the fuel drops of a high-speed engine. The number of drops of the above-computed limiting magnitude must be kept as small as possible.

Wöltjen's method affords the possibility of determining, in a relatively simple manner, the atomization attainable with a given nozzle and a predetermined pump pressure. We can obtain drops of any desired magnitude by changing one or the other or both of these factors. Thus the purely empirical method hitherto employed is replaced by a reliable, much simpler and cheaper method. The fact that the injection into the receiving

liquid takes place under conditions corresponding somewhat to those prevailing in an engine, affords a guaranty that the method is suited not only for comparing differently shaped nozzles, pumps, etc., but also has an absolute value, in that it closely approximates the actual atomization processes. Since Wöltjen chiefly used cold compressed air in his experiments, the conditions may have deviated somewhat from the actual conditions in a combustion chamber, due to the fact that the mass of the drops, injected at high velocity into the hot combustion air, is increased by the condensation of vapor on their surface or that their coefficient of resistance is affected by heat absorption. As to how far this is really the case can be determined by substituting hot nitrogen for the cold compressed air in the bomb.

With the determination of the size of the fuel drops much has already been accomplished since, on the basis of the known mass of the oil drop and its outflow velocity (as computed from the pump pressure), we can determine the length of its path, as likewise its velocity in the combustion chamber, whereby we must, however, assume that the mass and volume of the drop have not changed much during its journey. Nevertheless, we can always determine with sufficient accuracy how far from the nozzle the combustion chamber is swept. In this calculation we must always consider in what manner the velocity imparted to each drop as it leaves the nozzle is diminished under the in-

fluence of the friction of the air. While the resistance at low velocities increases according to Stokes' formula in proportion to the velocity, it changes at high velocities according to Newton's law of resistance approximately as the square of the velocity and proportional to the density of the air and the projection of the drop in the direction of motion. The upper limit of Stokes' formula depends on the Reynolds Number, which was determined from Lenhardt's experiments on the final velocity of freely falling water drops at $R = 50$. For fuel drops with diameters of about $30\,\mu$ and a kinetic viscosity of the air of $\nu = 0.15\ cm^2/sec.$, we obtain the lower limiting velocity for the applicability of the quadratic law of resistance at about 15 m (49.2 ft.)/sec. The outflow velocity of the fuel jet far exceeds this figure. Kuehn's calculations showed that, under the assumption of a drop having a diameter of $30\,\mu$ and an initial velocity of 250 m (820 ft.)/sec., it would still have a velocity of 30 m (98.4 ft.)/sec. after 0.00001 sec. in compressed air at 32 atm. The distance traversed during this time is at first 8 mm. After 0.0001 second, the velocity has fallen to about 4 m (13.1 ft.)/sec. (here Stokes' law of resistance must already apply), and the distance has increased to 16 mm (.63 in.). The result of this calculation is hardly affected by the fact that Newton's law of resistance does not apply in the vicinity of the velocity of sound, but that changes in the resistance, as already mentioned, take

place according to other laws. On taking this fact into consideration, we come to an interesting result, if, in working out the above-mentioned Büchner hypothesis, we give the jet a velocity greater than the velocity of sound, in order for it to have a high penetrating power with low resistance values. Although, according to what has been said, an increase in the coefficient of resistance can produce no further atomization, it does, however, considerably affect the attainable distance of travel of the drops, their deflection from the axis of the jet, and their variation in volume through the absorption of heat. The most favorable distribution of the atomized fuel jet would thereby be moved a little farther from the nozzle toward the center of the combustion chamber and consequently the conditions for the formation of a homogeneous mixture would be improved.

If, according to the above investigation, drops of 30 μ diameter show such a rapid decrease in their initial speed, the conditions will be still less favorable for smaller drops, a necessary requirement for high combustion speeds in vehicle engines. This would mean that the very finest fuel drops, which are the best adapted in size for favorable combustion, could not penetrate far enough into the compressed air in the combustion room, but would be mostly stopped near the nozzle. This would result in the accumulation of the drops near the nozzle, where there would not be enough oxygen for quick combustion. If this condition does not actually occur in the de-

scribed mass, it is due to the fact that the instituted considerations were based on the mass of only one drop and the assumption of an infinitely great mass of still air. In reality, however, we have to reckon with the kinetic energy of the whole fuel jet, which imparts to the air the momentum of all the drops, thereby making it the bearer of the fuel spray. The energy of the jet is imparted so much the better to the surrounding air, the closer the form in which the jet enters the air and the smaller the quantity of air, to which it must impart its velocity. The practical result of this argument for the constructor is therefore to conduct the fuel jet so that, in spite of the finest atomization, the component parts will be held together in as close a cone as possible, and that this cone can be utilized for accelerating or maintaining the desired air circulation. It is impossible to produce a perfect circulation throughout the whole combustion chamber, as in the air-injection method, simply through the kinetic energy of the fuel jet. Entirely apart from the fact that, in air injection, there is added to the mass of the fuel the not much smaller mass of the injection air, the latter is also greatly accelerated by its expansion. In order to obtain a like energy of flow in airless injection, pump pressures would have to be employed which would be multiples of those now customary. We would then obtain an extremely short injection period and explosive combustion, which would result in overstressing the already highly stressed driving gear.

If the requisite energy for distributing the fuel drops can not be fully obtained from the fuel jet, some other way must be found to replace the mixing action of the air-injection method. One of these means has already been mentioned. It consists in so conducting the atomization that, along with a number of small drops, a sufficient number of large drops will be produced, which, due to their greater momentum, will penetrate far into the combustion chamber and propagate the combustion begun by the small drops. This method may be suitable for stationary engines, although the production of drops of desired diameter, penetrating power, etc., can be but very imperfectly accomplished with the means now available. For high-speed vehicle engines, however, any method which delays the combustion process must be avoided as far as possible. This method will therefore be employed only to a limited degree and more attention will be concentrated in the direction of imparting such a motion to the combustion air as to make it the bearer of the fuel drops and thus distribute them throughout the combustion chamber. Even the first compressorless Diesel engines, namely, the Deutz displacer engines, had such a motion of the combustion air, whereby, however, more value was attached to the atomizing effect of the annular whirl than to the forced circulation of the air itself. Only of late has proper attention been given this question and has the endeavor been made so to direct the combustion air that it can be perfectly regulated and

brought, at a calculated velocity, to the place of combustion. Here it is accelerated by the fuel jet, carries away the fuel drops during the combustion period, drives the combustion gases ahead of it and furnishes fresh oxygen until the combustion is completed. The circumstance that, through the violent whirling of the air, more heat is imparted to the cylinder walls, is unimportant, in view of the fact that an increase in the revolution speed of vehicle engines is of much more importance than any possible better utilization of the fuel through less heat transmission.

In ignition-chamber engines the distribution of the fuel in the cylinder offers no great difficulty. On the one hand, it is comparatively easy to distribute the fuel sufficiently (especially when not too finely atomized) in the relatively small ignition chamber. On the other hand, the effect of the combustion gases, which are expelled at a high velocity from the ignition chamber into the cylinder, is to a certain degree comparable with the effect of the injection air. Even when (for example, in wide necks) the pressure difference between ignition chamber and cylinder is not very great, the burning gases will nevertheless undergo manifold changes in their direction of motion on their way to the working cylinder. These doubtless produce sufficient turbulence in the combustion chamber and consequently a good distribution of the fuel in the combustion air.

The production of the whirling motion for distributing the fuel is not so easy in mechanical-injection engines as in ignition-chamber engines. It is clear that the shape of the combustion chamber greatly affects the strength, direction and velocity of the air whirl sought. All recent endeavors have therefore been in the direction of forcibly mixing the fuel and combustion air by means of suitably shaped cylinder heads, pistons and ingenious arrangements of the nozzles. A combination of the annular-whirl method (Deutz-Brandis) with fine atomization was proposed by Bielefeld (Fig. 15). The fuel is injected under high pressure at the dead center in the finest state of atomization and is distributed by the air whirls, which are purposely generated as shown by the arrows in the figure ("Autotechnik," 1925, No.13, p.26). The annular concave shape of the cylinder cover and of the displacer head of the piston generate a whirling motion of the air which is maintained during the whole combustion period. It is worthy of note that in this construction the jet energy is not utilized to accelerate the revolving air, but that the fuel injected through a central nozzle in the form of a flat circular spray, strikes the air nearly at right angles and is carried away by it. The Krupp Diesel engine has a nozzle with several holes and guides the combustion air, by means of a circular "director" located at the inlet valve, in a circle around the cylinder axis (Fig. 16). By measuring the velocity of the circulating air current,

it was demonstrated that the most favorable combustion lay at a certain definite air velocity (about 8.5 m/sec. in the case investigated). At this velocity, which, in order of magnitude, agrees well with the values found by Hesselman in similar experiments, air molecules lying near the cylinder wall describe exactly a quarter circle. It may be assumed that the combustion is the most favorable when the air charge during the injection period describes the exact angle at the center which is included between two adjoining jets from the multiholed nozzle. Fig. 17 is a picture of the mushroom piston head of the Krupp Diesel engine at this favorable air velocity and plainly shows the strong scattering of the flames by the directed air flow. (The light spots on the piston head are deposits of zinc oxide due to keeping the fuel in galvanized-iron containers.) The more lightly built compressorless Diesel engine of the M.A.N. ("Maschinenfabrik Augsburg-Nürnberg,") in which the fuel is injected tangentially through two nozzles, likewise has a circular air flow and a disk-shaped combustion space (Fig. 18). This engine has the advantage of leaving the cylinder cover free from fixtures and enables the installation of such large inlet and outlet valves as to supply good air charges even at high revolution speeds. Above all, however, the kinetic energy of the fuel jets is here favorably utilized, because the turning moment of the injection energy efficiently supports the circular air flow. It is important for both nozzles, which are

supplied by one and the same pump, to have the same resistance. This is accomplished by equalizing the pipe resistances by means of exchangeable perforated plates in the distribution portion of the pipe. The high-speed two-stroke-cycle Diesel engine made by the "Hannoverische Waggonfabrik" (Fig. 19), likewise works with a circular motion of the air around the cylinder axis, due to the fact that the air, previously compressed in the crank case, is forced by a suitable disposition of the overflow ports to enter the cylinder in a tangential direction. Since the outlet valve is located in the cylinder head (contrary to the customary two-stroke arrangement), no disturbance is caused in the spiral motion of the air by the flow energy of the exhaust gases, so that the generated air whirl apparently persists throughout the whole duration of the injection and combustion process. The fuel is injected toward the cylinder axis through a nozzle located in the side of the cylinder head.

In addition to the above-described devices, the patent literature of recent years shows a large number of more or less practical proposals for automatically directing the combustion air. One idea recurs in many variations, namely, the guidance of the air column, driven by the piston toward the combustion space, by the archlike shape of the cylinder cover in such a way that its motion is reversed and given the same direction as the fuel jet. Fig. 20 represents a solution pro-

posed by myself, in which the combustion space is divided into two chambers, each having the shape of a paraboloid of revolution. The nozzle openings are located at about the foci of the paraboloids. As to whether the expected result, namely, a good distribution of the drops (produced by air whirls at the foci), will in fact come to pass can not be safely predicted, for the lack of experimental bases. The diminution of the air space to be swept by a nozzle, effected by dividing the combustion space into two symmetrical chambers, may nevertheless prove to be a suitable means for completely filling even larger combustion chambers with fuel drops.

On account of the importance of the question regarding the effect of the shape of the combustion space on the strength, direction and velocity of the desired air circulation, it would be desirable to investigate thoroughly, by systematic experiments, the laws of this whirling motion. In so far as the air flow can not be made visible by smoke, powdered wood, etc., it might be advisable to measure with a Pitot tube the air velocities in the compression chamber of an experimental engine throughout the whole cross section of the cylinder. It should thus be possible, by the systematic testing of the whole compression space to obtain, in a comparatively short time, a clear picture of the air circulation.

The complicated processes in the cylinder have been recently rendered accessible to direct visual observation. At the

Junkers works in Dessau, the combustion process in a compressorless Diesel engine was successfully photographed, through a quartz window in the cylinder head, by means of a stroboscope (Fig. 21). The pictures obtained (Figs. 22-24) plainly show the scattering of the flame under the influence of the motion of the air (which is circular in this case). The full-load pictures (Figs. 22-23) were taken in point of time, very near the dead center and shortly after the beginning of the ignition. The relatively small amount of fuel already injected was blown from the nozzle toward the right by the circularly moving air in the cylinder. The ignition had started at about the tip of the fuel jet and had then been quickly propagated backward to the nozzle (Fig. 22). After 0.0005 second, the flame had spread still farther (Fig. 23). On the other hand, Fig. 24 shows the final phase of a low-speed combustion. The correspondingly small low-speed flame had here already made one complete revolution in the combustion air and had been separated into several component flames ("V.D.I.," October 31, 1925, p.1372).

After the above detailed description of the injection and combustion process, something should also be said regarding the compression ratios suitable for vehicle engines, and on the shape of the combustion line in the indicator diagram. It is, of course, desirable to obtain a diagram which shows the greatest possible area at low maximum and expansion final pres-

sures. One way to do this, and which is best suited to the nature of vehicle engines, is to reduce the work of compression to a minimum, thereby making it easier to start the engine, whether by hand, compressed air or electric motor. In opposition to this, however, is the need of a high compression temperature for shortening the ignition delay and guaranteeing the ignition in a cold engine and at a low temperature of the inflowing air. The compression temperature must, of course, always be higher than the ignition temperature. For the correct adjustment of the former, it is therefore necessary to know the exact ignition temperature of the fuel to be used. Till very recently, the ignition point was restricted to the temperature at which self-ignition occurs in a uniform current of air or oxygen. Probably the most accurate values obtained by this method are the ones given in Table IV, as determined by Otto Alt, with the Krupp ignition-point tester (see N.A.C.A. Technical Memorandum No. 281, - Combustion of Liquid Fuels in Diesel Engine).

Table IV.

Ignition points (°C) in ignition-point tester and in engine.

Fuel oil	In oxygen stream in V_2A crucible	In engine	Hydrocarbon	In oxygen stream in V_2A crucible	In engine
Gasoline	255	385	Hexane	283	383
Kerosene	232 270-279	323 369-378			
Gas oil	275	350	Paraffin	246	340
Mexican fuel oil	276	375			
Paraffin oil	240	332			
Light tar oil	326	434	Phenol	590	744
Naphthalene oil	475	610	Naphthalene	557	705
Anthracene oil	425	550	Anthracene	493	630
Haltermann oil	471	605			
			Benzol	570	722
			Toluol	563	713
Vertical-furnace tar	465	597	Xylol	530	647
			Cymol	445	574

The ignition temperatures given for the engine are all probably too high, for the following reasons. All previous investigations of the self-ignition temperatures are inadequate because they were based on the assumption that the ignition

point is a simple conception. The investigations by Tauss and Schulte* demonstrated, however, that the same oil has different ignition points at different pressures and that these ignition points become lower with increasing pressure (that is, with increasing air density). In Fig. 25 the ignition points of several fuels are plotted against the pressures. According to the above-mentioned investigations, the assumption that the ignition temperature in the engine is about 100°C higher than in the open ignition-point tester can not be maintained. The question as to whether the ignition point is affected by the greater relative velocity between the air and fuel drops in the combustion chamber of the engine still requires experimental elucidation. It is possible that the increased transference of heat from the air to the fuel drops not only shortens the time required for ignition, but also lowers the ignition temperature itself. In any case, however, the compression temperature must, on account of the ignition delay, be about 50°C higher than the experimentally found ignition points of the fuels. The lowering of the compression temperature, due to leaks (piston play in cold engine), must be accounted for by a supplementary calculation.

On the basis of these assumptions, we can pass to the calculation of the compression pressure, whereby it must be borne

*"Ueber den Zündpunkt und Verbrennungsvorgang im Dieselmotor." Mitteil. Chem. Inst., Techn. Hochschule, Karlsruhe, No.2, 1924. See also N.A.C.A. Technical Memorandum No. 299.

in mind that a vehicle engine should be able to function under the most unfavorable conditions. If we assume that in winter it must be able to start at an air temperature of -10°C (14°F) and if we adopt 300°C (i.e., the mean value between the ignition temperatures of Alt and of Tauss and Schulte) as the lower limit for the ignition temperature of gas oil, we then have, on the assumption of adiabatic compression,

$$p_2 = \sqrt[\frac{1-m}{m}]{\frac{T_c}{T_a}}\, p_1 \qquad (6)$$

If $m = 1.3$ is chosen as the exponent of the compression and $p_2 = 0.95$ atm. as the initial pressure (for engines up to about 1200 R.P.M.), we have $p_2 = 28$ atmospheres.

If we consider that the compression line approaches the isotherm at low revolution speeds, due to the cooling effect of the cylinder walls, and that, above all, the unavoidable piston leaks, which are more unfavorable in the small bores of vehicle than in larger engines, still further reduce the final compression pressure, we will not then be inclined to go below the calculated value. If, however, this generally happens, it can only be made possible by not requiring the engine to start at such low external temperatures without some special auxiliary device and by limiting ourselves to calculating with air temper-

atures of at least 10°C (50°F). At lower temperatures, therefore, the combustion air must be previously heated by electricity, or the start must be made with the aid of an ignition cartridge or easily inflammable paraffin oils. The great effect of previously heating the air is shown by the calculation, which, at an inflowing-air temperature of 0°C (32°F) under the same conditions as above, gives a final compression pressure of 24.9 atm., or 20.7 atm. at 10°C (50°F).

In the warm running condition of the engine, the inflow temperatures of the air increase greatly and reach 80-90°C at full load. In most cases a final compression pressure of 25 atm. is regarded as sufficient to assure the starting of the engine without any special auxiliary device. We can go to this extreme limit, because the cooling effect of the injection air is eliminated. At the small safety factor, the greatest possible tightness of the valves and pistons is a prerequisite condition.

The further functioning, after compression, depends entirely on the method of introducing the fuel. The pressure line of the indicator diagram will ascend, move horizontally, or descend, according to the timing and duration of the injection. Its shape is therefore chiefly determined by the nature of the nozzle or by the pump-pressure diagram. If the fuel is injected very quickly at about 40-45° before the upper dead center, there is an explosion similar to the one in a carburetor

engine, as shown by the pointed indicator diagrams. Injection at 25-27° before the dead center produces, on the other hand, according to Heidelberg, a constant-pressure combustion. If the mean pressure is not allowed to fall below 5.5 atm., the usual value for modern carburetor engines (a necessary condition for vehicle engines, due to spatial considerations), maximum pressures of about 80 atm. are produced in a purely explosive functioning. Such a pressure increase is impossible, however, due to the great stressing of the working parts and the consequent necessary increase in their weight. A constant-pressure combustion, similar to that obtained by air injection, is indeed easily attainable in compressorless Diesel engines, but experiments have demonstrated that this form of the combustion line is neither practical nor economical. The slight ignition delay, unavoidable in airless injection, can best be accounted for by a short constant-volume combustion, followed by constant-pressure combustion (Fig. 26). This kind of mixed combustion, which occurs chiefly in engines with mechanical injection, produces a maximum pressure of 45-50 atm., on the assumption of a mean electric pressure of about 5.5 atm., and a compression of 25 atm.

The maximum pressure of the working diagram therefore determines the size of the connecting rods, while the dimensions of the crank shaft are determined by the tangential-pressure diagram. In a connecting rod of circular cross section, the

diameter of the rod is proportional to the square of the maximum pressure and its weight is directly proportional to the pressure. These values throw some light on the weight relations of a Diesel vehicle engine in comparison with a carburetor engine and partially determine its structure. In comparison with an ordinary motor-truck engine with a final compression pressure of 5.5 atm., and a maximum combustion pressure of about 27 atm., we obtain, with a Diesel engine of like mean pressure and like revolution speed, an increase of about 60% in the weight of the connecting rods and of about 35% for the crank shaft. Dr. W. Riehm, in his lecture on "High-Speed Diesel Engines for Vehicles,"* delivered before the Association of German Engineers at Augsburg, in 1925, gives considerably smaller values for the increase in the weight of these parts. His assumption that the diameter of the connecting rods (having a perfectly round cross section) should be proportional to the fourth root of the maximum pressure, is based, however, simply on their resistance to buckling and not on their admissible compressive stress. In the dimensioning of the other engine parts, the increasing of the combustion pressure does not necessitate so great weight increases, because the pistons, cylinders, crank cases, etc., have more than sufficient strength, for reasons pertaining to the casting and finishing. The fuel pumps, which take the place of the magnetos and carburetors,

*"Schnellaufende Dieselmotoren für Fahrzeuge," published in "V.D.I.," August 29, 1925, pp. 1125-1131.

are about twice as heavy as those parts, so that, all in all, a Diesel engine weighs 30-35% more than a carburetor engine of the same power and revolution speed.

There is hardly anything to be said on the structure of vehicle Diesel engines, since it is fully based on modern carburetor engines. Of course the different operating conditions necessitate corresponding modifications in the various engine parts. Since the reliability of the ignition depends largely on the final compression pressure, much attention should be given to making the valves and piston as tight as possible. While the former can be protected from harmful heat stresses by the use of suitable material and by cooling the valve seats, the piston, in addition to stronger dimensioning of the piston head and piston pin, can be made longer and be provided with a greater number of rings. The use of light-metal pistons is not desirable, at least so long as there is no harmful heat accumulation with cast-iron pistons. The piston play, which must be rather large with aluminum, on account of the high coefficient of heat expansion, expresses itself, in a cold engine, in leaks which may impair the starting ability. Moreover, the larger dimensioning of the piston pin and the avoidance of inadmissible surface pressures in the piston-pin bearings involve certain structural difficulties.

The fact that, due to irregularities in the pump, faulty atomization, etc., misfires occur and that subsequently the

the combustion of greater quantities of fuel with excessive pressure increase can follow, raises the question of providing a safety valve for every cylinder. Entirely aside from the fact, however, that there is seldom room enough for such a valve in the cylinder cover, I believe that the danger of a sudden strong pressure increase, which could be disastrous for the engine, should not be over-rated, because the condensed, unburned fuel in the cylinder would be only slightly consumed in a subsequent ignition, while most of it would pass unburned into the exhaust.

The impossibility of cranking the engine against the full compression pressure necessitates the use of reduction gearing. It is generally customary to raise the starting valve either by a special lever or, better, by the axial shifting of a cam shaft provided with auxiliary cams. If these auxiliary cams are conical, the valves in starting can be partially or completely opened at will. The energy stored up in the flywheel in starting under diminished pressure then facilitates throwing the engine into gear against the full compression pressure. The compressed-air method of starting, customary with stationary engines, can at most be used only on such motor vehicles as carry compressed-air cylinders for special reasons, such as for air brakes and gear shifting. Since an electric generator is essential on a modern motor vehicle, the engine

is generally started by a rather large starting motor. Provision must also be made for starting by hand in an emergency. (To be followed by Part III.)

Translation by Dwight M. Miner,
National Advisory Committee
for Aeronautics.

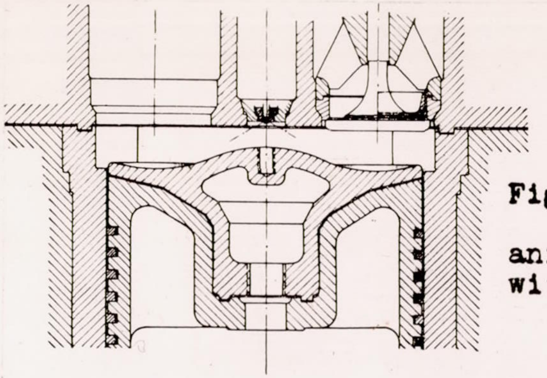


Fig.12 Combustion space of a Krupp mechanical injection engine with mushroom piston head.

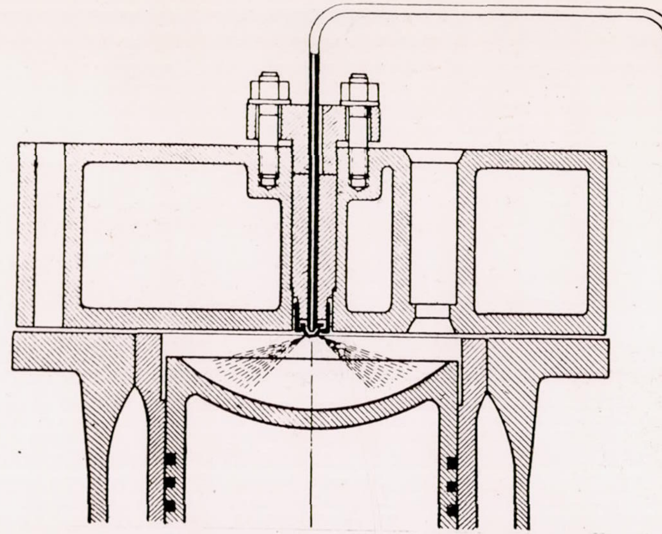


Fig.11 M.A.N. engine with concave piston head.

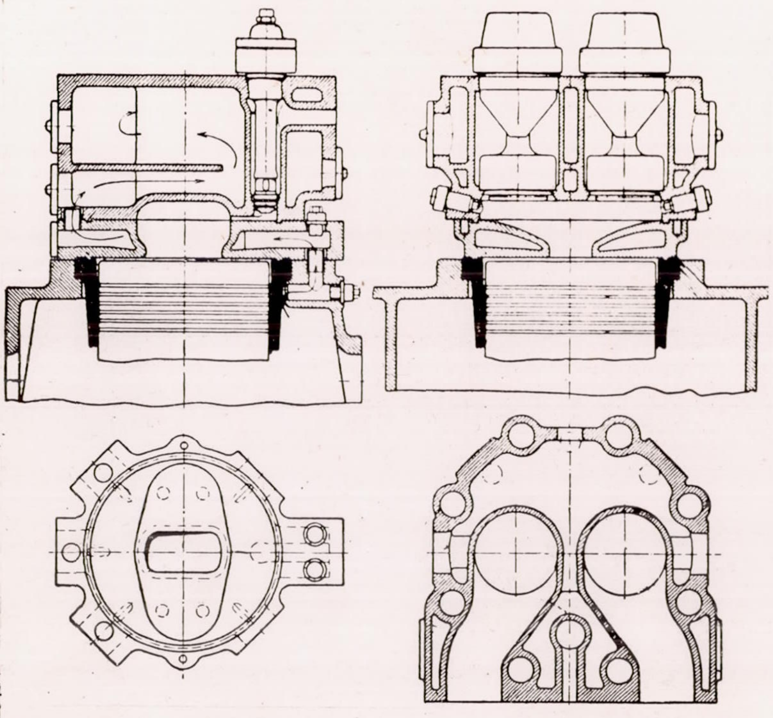


Fig.13 Combustion space of the Falk oil engine, similar to Price type.

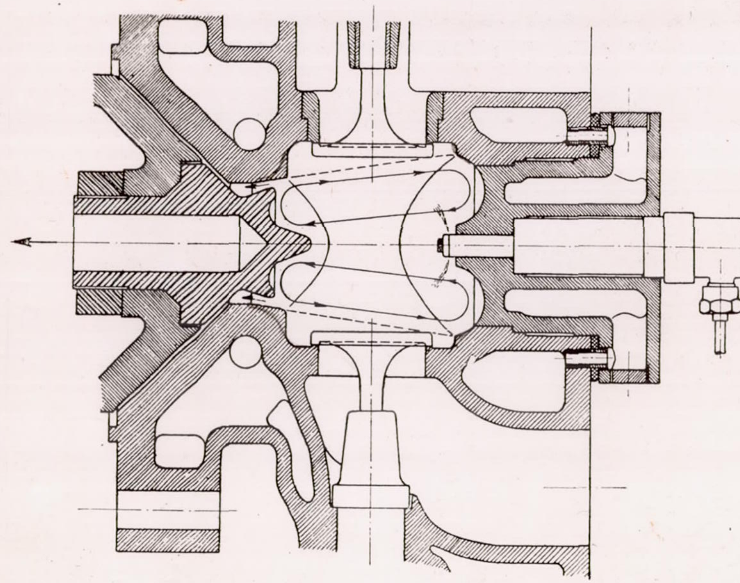
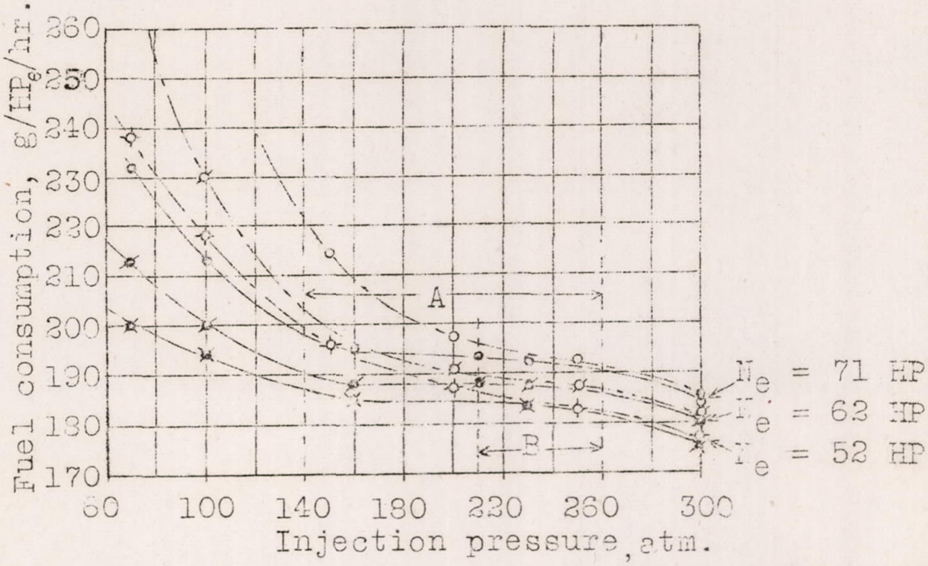


Fig.15 Combustion space proposed by Bielefeld.

———— Nozzle with 3 holes of 0.5mm diameter. Jet angle 90^0.

— - — - - Nozzle with 1 hole of 2mm diameter.



A = Constancy range of three-hole nozzle.
B = Constancy range of one-hole nozzle.

Fig.14 Constancy of various nozzles (according to Heidelberg.)

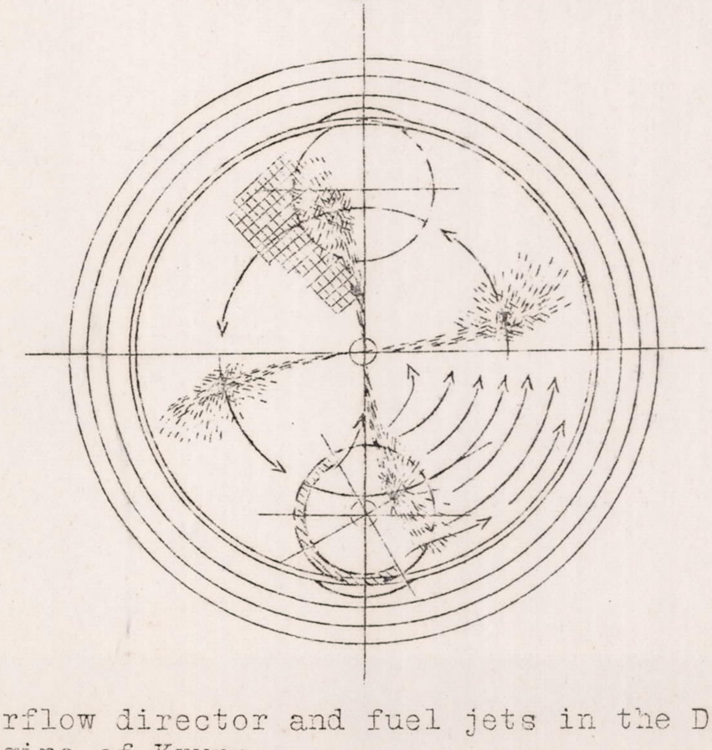


Fig.16 Airflow director and fuel jets in the Diesel engine of Krupp.

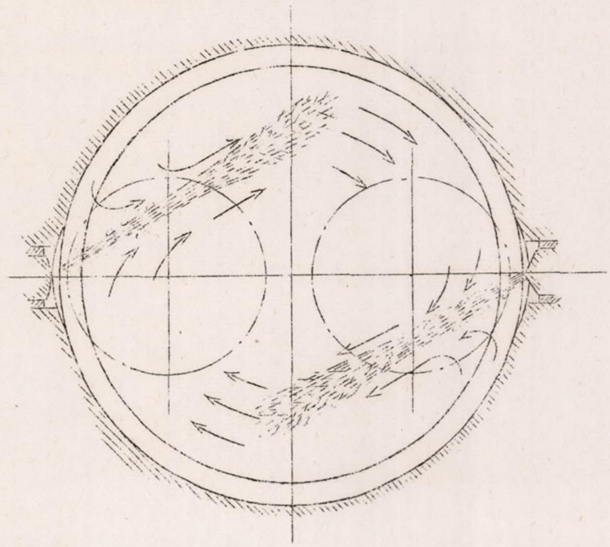


Fig.18 Airflow directors and fuel jets in the M.A.N. stationary Diesel engine.

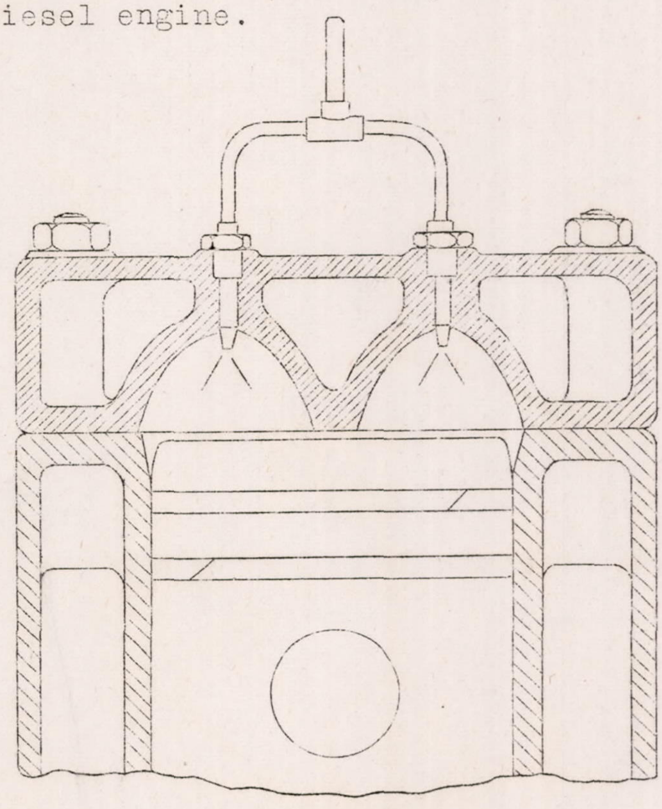


Fig.20 Combustion space proposed by Hausfelder.

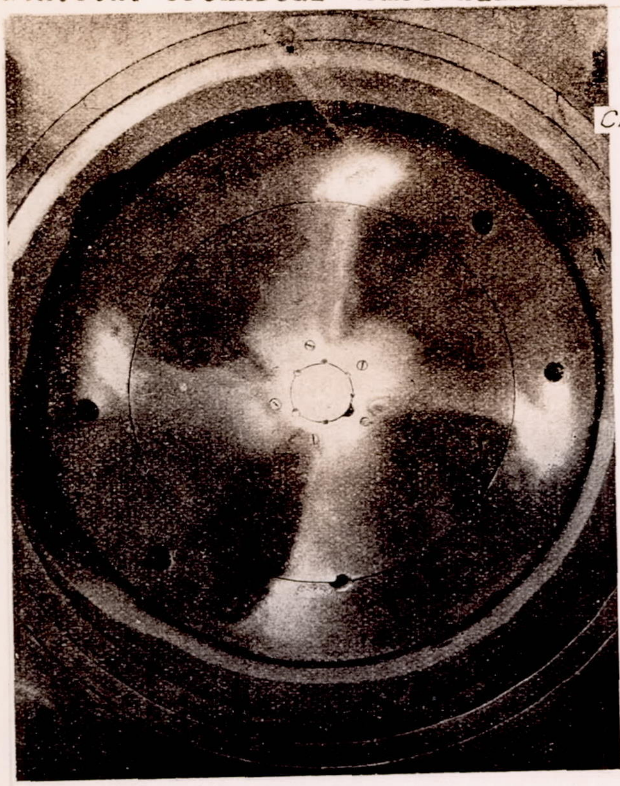


Fig.17 Mushroom piston head of Krupp Diesel engine.

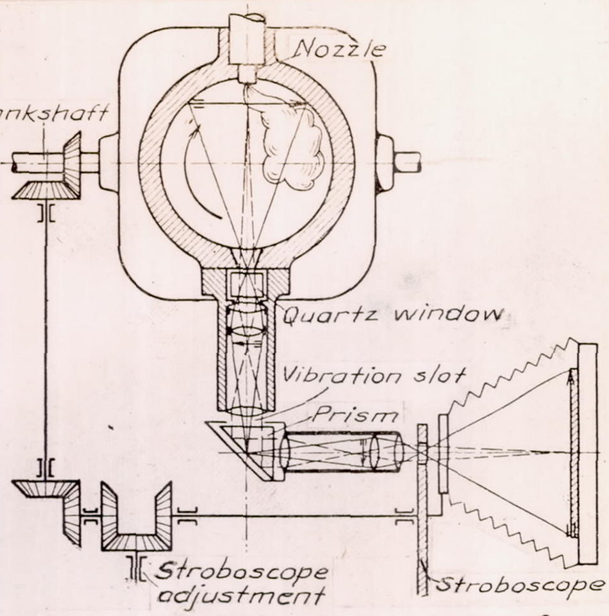


Fig.21 Junkers apparatus for photographing the combustion process.

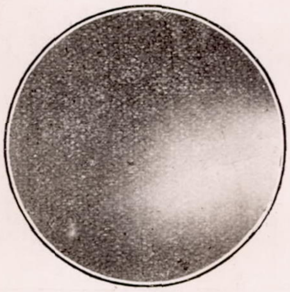


Fig.22 3.3° before dead center

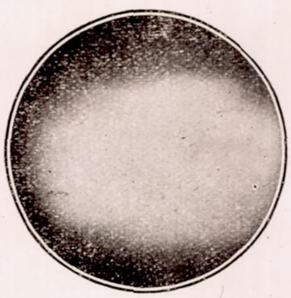


Fig.23 1.7° before dead center

Fig.24 15° after dead center

Fig.22,23 & 24 Flame pictures (Junkers)

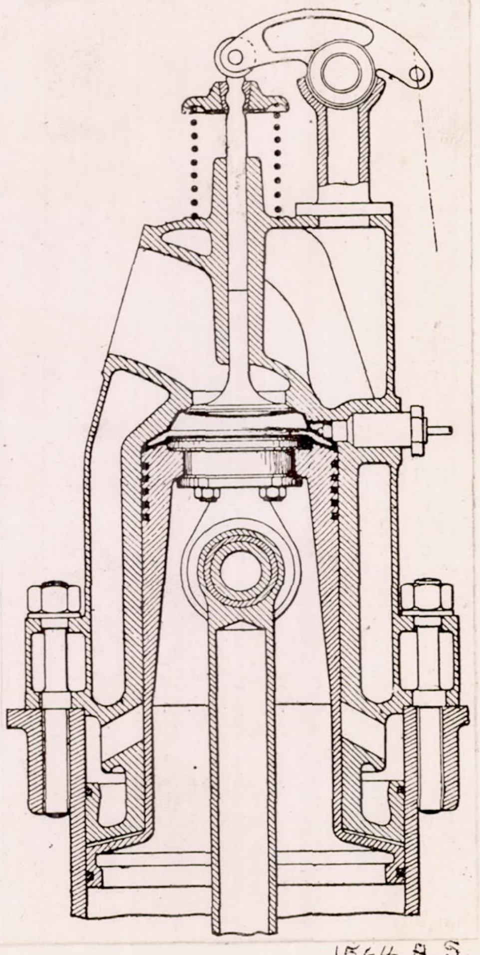


Fig.19 Diesel engine of Hannover "Waggonfabrik.

1564 A.S.

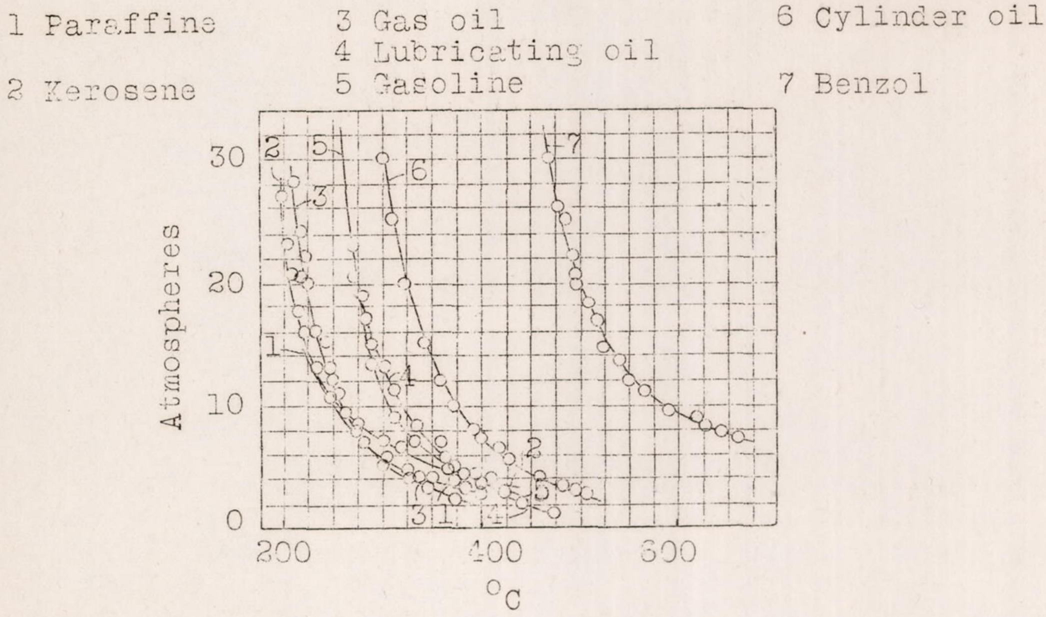


Fig.25 Ignition temperatures plotted against pressures. (Tauss and Schulte)

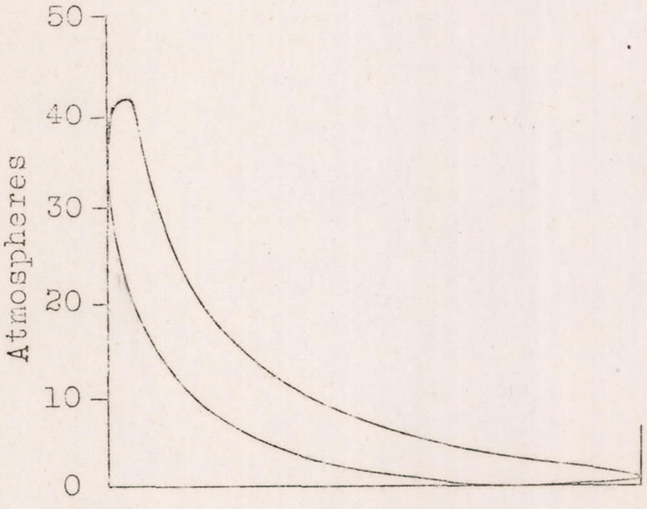
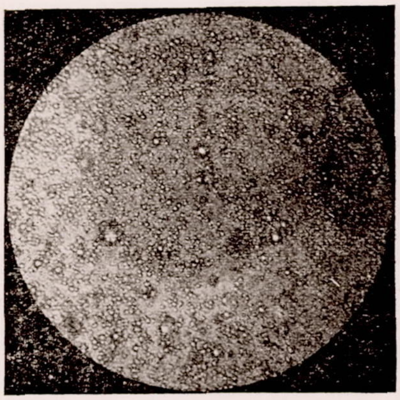
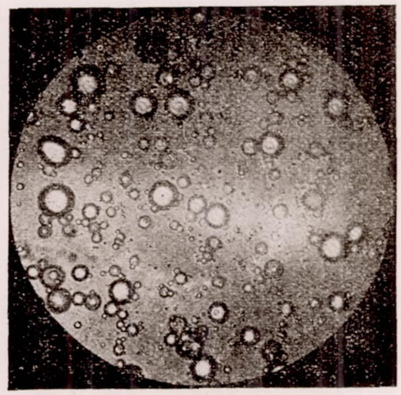


Fig.26 Normal indicator diagram of an airless-injection Diesel engine.

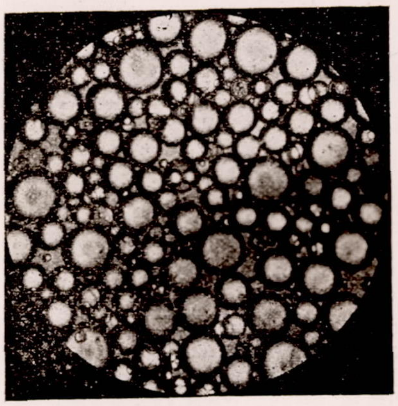
Plate I (Air injection). Compression pressure, 30 atm.



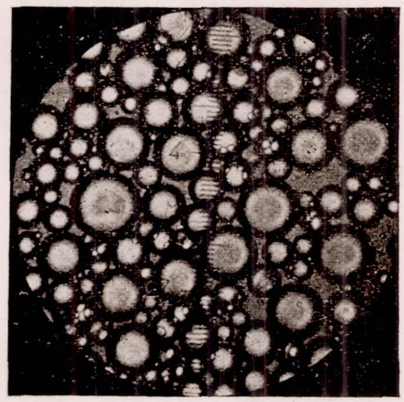
a) pe = 75 at
dm = 4,37 μ



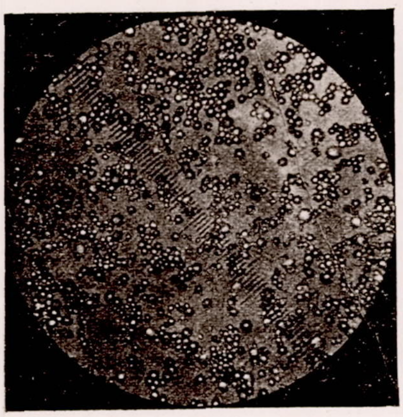
b) pe = 65 at
dm = 13,75 μ



c) pe = 50 at
dm = 17,5 μ

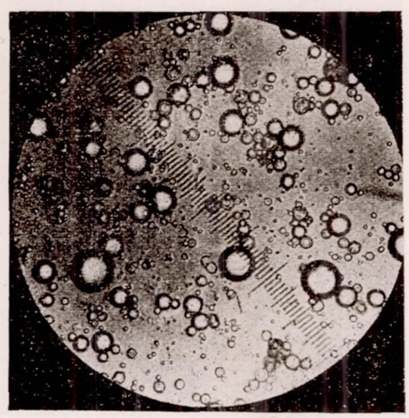


d) pe = 40 at
dm = 25 μ



e) dm = 4,5 μ

Cow-milk cream



f) pe = 150 at
dm = 13,75 μ

Airless injection with dispersing nozzle.

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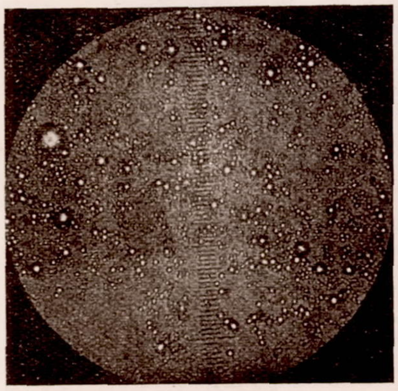
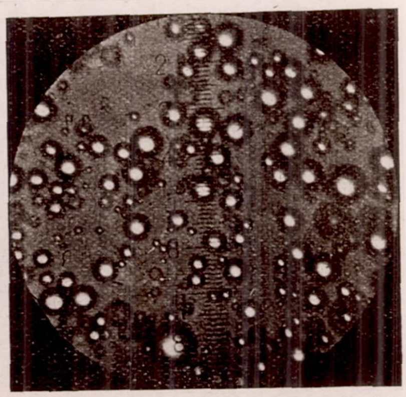
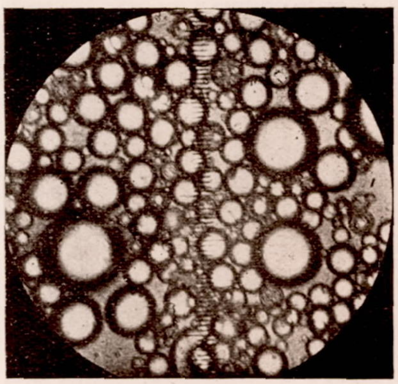
Plate II (Airless injection). Compression pressure, 30 atm.



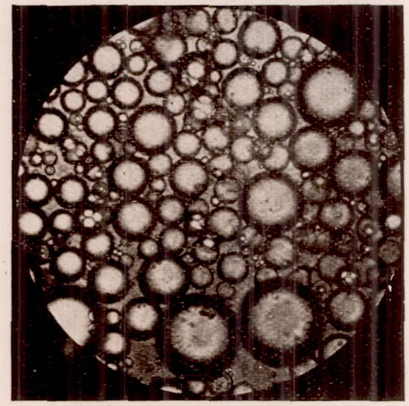
g) p_e = 300 at
d_m = 4,37 μ



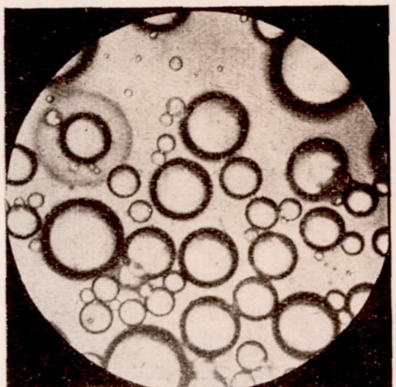
h) p_e = 250 at
d_m = 13,75 μ



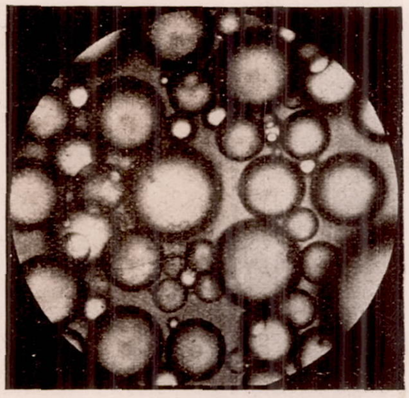
i p_e = 200 at
d_m = 20 μ



j) p_e = 150 at
d_m = 26,25 μ



k) p_e = 100 at
d_m = 33,75 μ



l) p_e = 50 at
d_m = 40 μ

1565 A.S.